

Optimization of a Passive Vehicle Suspension System for Ride Comfort Enhancement with Different Speeds Based on DOE Method

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Abstract— This paper reports on an investigation to determine the spring and damper settings that will ensure optimal ride comfort of vehicle in different speeds using Design of Experiment Method (DOE). The extent to which the ride comfort optimal suspension settings vary for roads of different roughness and varying speeds and the levels of ride comfort that can be achieved, are addressed. Optimization is performed with the DOE method with 7 DOF modeled for speeds ranging from 60 to 90 km/h. Results indicate that optimization of suspension settings using different road roughness and specified range of speed improve the ride comfort. It was found that ride comfort is most sensitive to a change in rear spring stiffness.

Index Terms— suspension settings, ride comfort, optimization, Design of Experiment, Vehicle

I. INTRODUCTION

Ride comfort is one of the most critical factors to evaluate the automobile performance and has been an interesting topic for researchers for many years. Automobile designers give an abundant attention to the isolation of vibrations in the car, in order to provide a comfortable ride for the passengers. Despite of all progresses in vehicle design profession, the determination of vibration comfort is still a subject of conflict between automobile designers, and so far no common standard has been developed on this matter [1]. Two methods, i.e. computer simulation and road experiment, are used to investigate ride comfort. Computer simulation method [2] is based on the mathematical model of the vehicle vibration and power spectral density (PSD) of road surface. Duncan [3] used FEM models for suspension system to study the ride quality and using numerical optimization methods recommended some modifications on suspension system. Tamboli and Joshi [4] tried to produce real data from road real profile and optimize the suspension system under real road inputs. One of the complete works in this area is the simulation done by Gobbi at Fiat Company [5]. He used a car model that was developed in ADAMS using real road inputs, and constructed an optimization scheme that was successfully implemented to their car. Karnopp considered the vehicle model as a single degree of freedom quarter model with active and passive damping and linear stiffness subjected to white noise base velocity excitation [6]. Macvean and Robson

have investigated stationary response to profile imposed excitation with randomly varying traverse velocity and showed that the change in velocity makes a proportional contribution to the response [7]. Hac and Youn showed that incorporation of a time delay between the front and rear axles in controller design improves the dynamic behavior when road excitation is simulated by white noise and vehicle velocity is constant [8]. Elbehiery and Karnopp have optimized suspension system parameters for five types of suspension systems to obtain constant root mean square suspension deflection [9]. And finally the valuable work of Christensen, Stein and Ferris in ARC at University of Michigan must be mentioned. They used sensitivity analysis and DOE optimization techniques on an ADAMS car model to optimize car driving and ride performance [10]. In this investigation, the 7DOF vehicle model has been modeled and analyzed. To study the ride comfort of the vehicle, vertical acceleration and pitch angle have been calculated and optimized. For this case, the coefficients of front and rear (F/R) springs and dampers out of many other factors such as suspension geometry, mount and joint characteristics have been varied in the simulation. It is because of the dominant role of F/R springs and dampers. Finally, DOE method is applied to decide the best values of selected parameters to fulfill the best ride comfort. A generic simulation procedure is illustrated in Fig. 1.

II. RIDE COMFORT

The ride of a vehicle is the heaving, pitching and rolling motion in forced vibration caused by road roughness. The purpose of the suspension is to minimize the discomfort of the passengers, which obviously involves a minimization of some measure of the vehicle body motion, by choice of the springs and dampers as shown in Fig. 2 [11]. Since passenger comfort is not a quantitative parameter, majority of research efforts have been focused on devising a qualitative measure of passenger comfort (also referred to as human response to vibration). Passenger comfort principally depends on magnitude and direction of acceleration and the frequency of vibrations acted on his body. Four methods to objectively evaluate ride comfort are used throughout the world today. The ISO 2631 standard [12] is used mainly in Europe and the British standard BS 6841 [13] in the United Kingdom. Germany

and Austria use VDI 2057 [14], while average absorbed power or AAP [15] is used by the United States of America and by NATO in the NATO reference mobility model (NRMM). Ride behavior may be analyzed in the time domain or in the frequency domain. Time-domain analysis predicts positions, velocities and accelerations as functions of time. Frequency domain analysis predicts the characteristics as functions of frequency, e.g. the transmissibility, hence revealing resonances, etc.

III. DOE METHOD

Design of Experiments or DOE is a collection of procedures and statistical tools for planning experiments and analyzing the results. In general, the experiments measure the performance of a physical prototype, the yield of a manufacturing process, or the quality of a finished product. Experiments are performed by investigators in virtually all fields of inquiry, usually to discover something about a particular process or system. Literally, an experiment is a test. More formally, an experiment can be defined as a test or series of tests in which purposeful changes are made to the input variables of a process or system so that one may observe and identify the reasons for changes that may be observed in the output response [16]. The process or system can be represented by the model shown in Fig. 2. It can be usually visualized the process as a combination of machines, methods, people, and other resources that transforms some input (often a material) into an output that has one or more observable responses.



Fig 1. A generic computer simulation procedure

Some of the process variables X_1, X_2, \dots, X_p are controllable, whereas other variables Z_1, Z_2, \dots, Z_q are uncontrollable (although they may be controllable for purposes of a test). The objectives of the experiment may include the following:

- Determining which variables are most influential on the response y
- Determining where to set the influential X 's so that y is almost always near the desired nominal value
- Determining where to set the influential X 's so that variability in y is small.
- Determining where to set the influential X 's so that the effects of the uncontrollable variables Z_1, Z_2, \dots, Z_q are minimized.

As can be seen from the foregoing discussion, experiments often involve several factors. The main goal of experimental design is determining variations that have more effect on responses. After that, optimization of effective parameters could be done. There are two aspects to any experimental problem: the design of the experiment and the statistical analysis of the data. These two subjects are closely related because the method of analysis depends directly on the design employed. The three basic principles of experimental design are replication, randomization, and

blocking. By replication we mean a repetition of the basic experiment. Randomization is the cornerstone underlying the use of statistical methods in experimental design. By randomization we mean that both the allocation of the experimental material and the order in which the individual runs or trials of the experiment are to be performed are randomly determined. Blocking is a design technique used to improve the precision with which comparisons among the factors of interest are made.

Guidelines for Designing an Experiment are as follows:

- Recognition of and statement of the problem.
- Choice of factors, levels, and ranges.
- Selection of the response variable.
- Choice of experimental design.
- Performing the experiment.
- Statistical analysis of the data.
- Conclusions and recommendations.

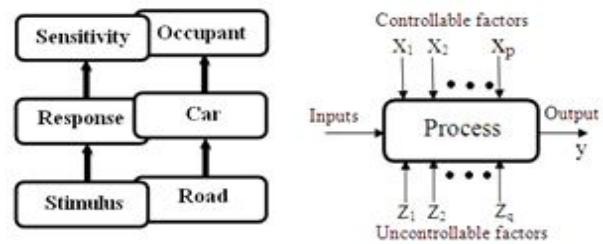


Fig 2. General model of a process.

IV. VEHICLE MODEL

A full car vibrating model has seven DOF according to Fig. 4 with the equations of motion (1) to (7). The vehicle body is represented by three degree-of-freedom rigid cuboids with mass M . The heave, pitch and roll motions of the sprung mass are considered. The four unsprung masses (front-left, front-right, rear-left and rear-right) are connected to each corner of the rigid cuboid. It is assumed that the four unsprung masses are free to bounce vertically. Shock absorbers are far from being linear viscous dampers. In fact, most automotive shock absorbers are unsymmetrical, with a damping which is larger in the rebound stroke. The suspensions between the sprung mass and unsprung masses are modeled as nonlinear spring and nonlinear dampers elements, while the tires are modeled as nonlinear springs with viscous damping. It is assumed that the modeled nonlinear spring of suspension has the curves as shown in Fig 4 and 5.

The parameters of the vehicle model which are used in the numerical study are shown in Table 1.

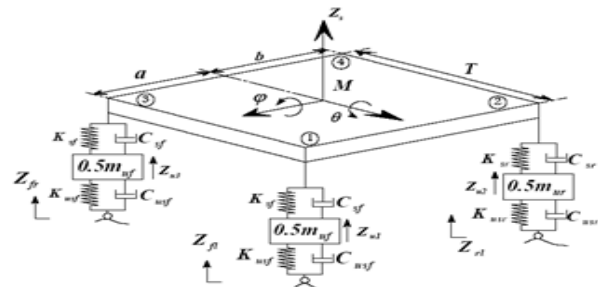


Fig 3. A full 7DOF vehicle model.

$$M\ddot{Z}_s + F_{1s} + F_{1c} + F_{2s} + F_{2c} + F_{3s} + F_{3c} + F_{4s} + F_{4c} = 0 \quad (1)$$

$$I_\phi \ddot{\phi} + \frac{T}{2} \cos \phi (F_{1s} + F_{1c} + F_{2s} + F_{2c} - F_{3s} - F_{3c} - F_{4s} - F_{4c}) = 0 \quad (2)$$

$$I_\theta \ddot{\theta} - a \cos \theta (F_{1s} + F_{1c} + F_{3s} + F_{3c}) + b \cos \theta (F_{2s} + F_{2c} + F_{4s} + F_{4c}) = 0 \quad (3)$$

$$0.5m_{uf} \ddot{Z}_{u1} - (F_{1s} + F_{1c}) + K_{uf} (Z_{u1} - Z_{f1}) + C_{uf} (\dot{Z}_{u1} - \dot{Z}_s) = 0 \quad (4)$$

$$0.5m_{ur} \ddot{Z}_{u2} - (F_{2s} + F_{2c}) + K_{ur} (Z_{u2} - Z_{r1}) + C_{ur} (\dot{Z}_{u2} - \dot{Z}_s) = 0 \quad (5)$$

$$0.5m_{uf} \ddot{Z}_{u3} - (F_{3s} + F_{3c}) + K_{uf} (Z_{u3} - Z_{fr}) + C_{uf} (\dot{Z}_{u3} - \dot{Z}_s) = 0 \quad (6)$$

$$0.5m_{ur} \ddot{Z}_{u4} - (F_{4s} + F_{4c}) + K_{ur} (Z_{u4} - Z_{rr}) + C_{ur} (\dot{Z}_{u4} - \dot{Z}_s) = 0 \quad (7)$$

Where for $i=1:4$ the amount of F_{is} vs. displacement And F_{ic} vs. velocity is shown in Fig. 5, 6 numerically.

$$\Delta_1 = Z_s - a \sin \theta + T/2 \sin \phi - Z_{u1} \quad (8)$$

$$\Delta_2 = Z_s + b \sin \theta + T/2 \sin \phi - Z_{u2} \quad (9)$$

$$\Delta_3 = Z_s - a \sin \theta - T/2 \sin \phi - Z_{u3} \quad (10)$$

$$\Delta_4 = Z_s + b \sin \theta - T/2 \sin \phi - Z_{u4} \quad (11)$$

TABLE I NUMERICAL VALUES OF THE SYSTEM PARAMETERS

Parameter	Value
Sprung mass, M	1295 kg
Roll axis moment of inertia, I_ϕ	200 kg.m ²
Pitch axis moment of inertia, I_θ	500 kg.m ²
Front unsprung mass, m_{uf}	50 kg
Rear unsprung mass, m_{ur}	50 kg
Tire spring stiffness, K_{uf} , K_{ur}	190,000 N/m
Damping coefficient of tire, C_{uf} , C_{ur}	500 N.s/m
Length between sprung mass and front vehicle, a	1.233 m
Length between sprung mass and rear vehicle, b	1.327 m
Width of sprung mass, T	1.55 m

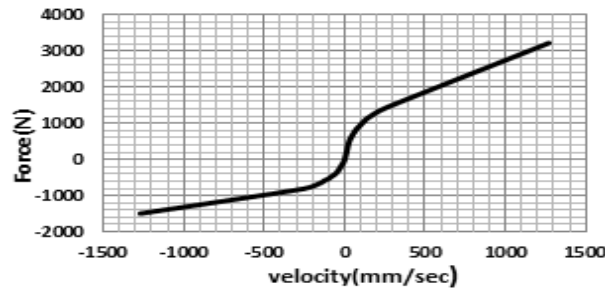


Fig. 4. Damper characteristic curve.

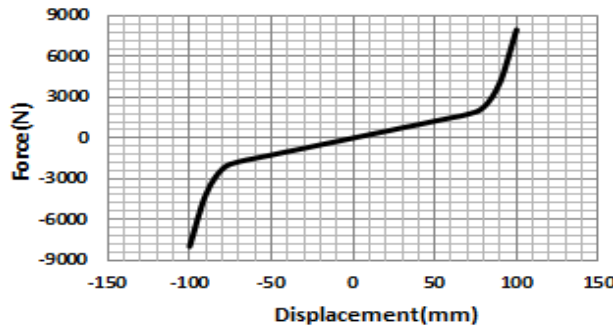


Fig 5. Spring characteristic curve

V. ROAD PROFILE

Knowledge of the excitation due to motion on uneven road is important for the study of riding comfort. Road excitation could enhance the impact to the automotive chassis and increase the applied vibration to the passengers. Because such excitation cannot be studied with a deterministic approach, the methods used for random vibrations must be applied. A number of studies have been devoted to characterizing the road profiles experimentally and interpreting the results statistically. For a single wheel track, the Sayers Roughness Model assumes that the power-spectral density (PSD) of the displacement (elevation) of a

road profile, G_d , is a function of wave number, ν , given by the equation (12) as stated in [17]:

$$G_d(\nu) = G_s + \frac{G_v}{(2\pi\nu)^2} + \frac{G_a}{(2\pi\nu)^4} \quad (12)$$

Therefore, it is assumed that roughness comes from three components. Each is obtained from an independent source of white noise, that is, random numbers:

a. The first component, with amplitude G_e , is white-noise elevation.

b. The second, with amplitude G_s , is white-noise slope (velocity) that is integrated once with respect to time.

c. The third, with amplitude G_a , is white-noise acceleration that is integrated twice with respect to time.

The constants of the Sayers Roughness Model are listed in Table 2. Assuming a constant vehicle velocity of 28 m/s and referring to Table 2, the excitation of tires caused by road surface versus time is shown in Fig. 6. It is clear that there is a phase difference between front and rear tires excitation depending on the vehicle velocity and an insignificant difference between left and right tires.

TABLE II. VALUES FOR SAYERS ROUGHNESS MODEL

parameter	G_s	G_v	G_a
unit	$\left(\frac{m^3}{cycles} \times 10^{-6}\right)$	$\left(\frac{m}{cycles} \times 10^{-6}\right)$	$\left(\frac{1}{(m \times cycles)} \times 10^{-6}\right)$
value	0.1	20	0.1

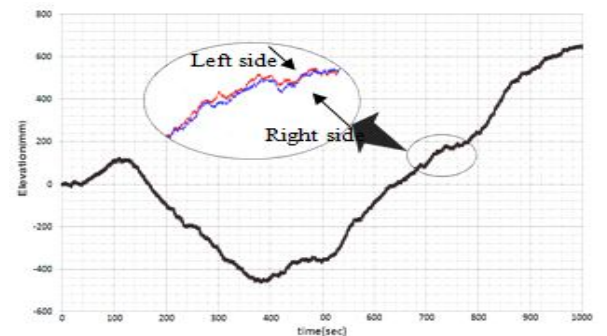


Fig. 6. Time history of road irregularities and surfaces

VI. DETERMINING OPTIMAL SUSPENSION SETTINGS

Several questions have to be addressed when determining the optimal suspension settings. The first is to what extent does the optimal suspension settings vary if a road of given roughness is traversed at different speeds. Secondly, what levels of ride comfort can be achieved in these cases? Another question at hand is if an off-road vehicle travels over roads of different roughness with a suspension setting optimized for one road at a specified speed, what is the effect on ride comfort? Furthermore, what is the effect if a vehicle travels at a specified speed over a particular road profile with the suspension setting optimized for a different speed? Also how much will the ride comfort be affected and how much can the suspension settings vary so that the ride comfort is still within 5% of the optimal suspension value?

VII. OPTIMIZATION DESIGN OF RIDE COMFORT BASED ON DOE METHOD

6.1. DESIGN VARIABLES

Due to the fact that the geometry of a suspension system is optimized based on handling and road holding, the geometry of a suspension system has not been taken into account as a design variable. In fact, ride quality mostly depends on the characteristics of rear and front springs and dampers. In this study, the 7DOF vehicle model as stated in (1) to (7) with numerical parameters as Table 1 is subjected to the road surface (14) in an asphalt profile according to Table 2. The vertical acceleration and pitch angle of the auto-body is calculated to be observed the effect of those two mentioned parameters in the forward and rearward springs and dampers. In Fig. 7, the effect of variation of such characteristics on ride quality is illustrated. The optimization procedure is done according to a basic curve of spring and damper characteristics and a correction factor defined as the ratio of corrected to basic spring and damper coefficient. Four design parameters chosen for experiment are shown as vector v :

$$v = \{f_{sf}, f_{sr}, f_{cf}, f_{cr}\}$$

where: f_{sf}, f_{sr} are front and rear spring correction factors

and f_{cf}, f_{cr} are front and rear damper correction factors.

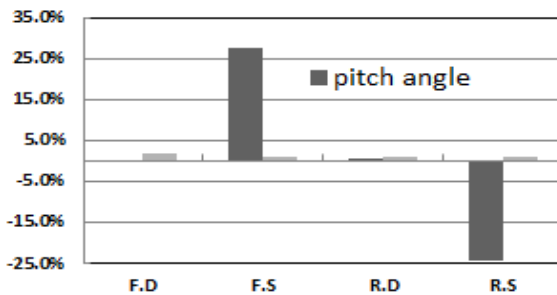


Fig.7. The effect of spring and damper characteristic on ride qualities

Design variable are limited to the ranges defined by the bounds shown in Table 3. In practice, these ranges reflected technical abilities to manufacture theme in usual passenger cars.

TABLE III. CORRELATION FACTOR'S BOUND LIMIT

Design variable	f_{sf}	f_{sr}	f_{cf}	f_{cr}
Lower bound limit	0.4	0.6	0.8	0.8
Upper bound limit	1.4	1.6	1.2	1.2

6.2. OBJECTIVE FUNCTION

Since ride comfort is a qualitative subject, suitable criteria shall be defined to quantify it. The objective function which is used here is designed based on the combination of vertical acceleration RMS (Root Mean Square) and pitch angle as:

$$S = w_1 \frac{a_z_{RMS}}{a_z_{MAX}} + w_2 \frac{\theta_{RMS}}{\theta_{MAX}}$$

where w_1, w_2 are weight factors and a_z, θ are vertical acceleration and pitch angle. There is no defined criterion in determination of weight factors. Vehicle industry professionals have offered some values for weight factors [1]. Here, , are chosen based on the range of their advice.

6.3. OPTIMIZATION RESULT

In this research, the optimization is accomplished in the speed range of 60 to 90 Km/h based on DOE method. The design variables are v as stated in Eq. (13) and the objective function is S as expressed in Eq. (14). Here, the final goal of optimization is estimating design variables. In this case, optimum values of design variables have been calculated and are shown in Table 4. Increasing stiffness of rear spring and damper and decreasing stiffness of front spring and damper cause to improve ride comfort results as may be observed in Table 4.

TABLE IV. OPTIMUM VALUE OF DESIGN VARIABLES

Design variable	f_{sf}	f_{sr}	f_{cf}	f_{cr}
Optimized value	0.87	1.19	0.825	1.38

Vertical acceleration and pitch angles of the model (objective functions) are compared before and after optimization and the results may be seen in Fig. 8 (a and b) for velocity of 70 km/hr. It is done to show that the optimization works acceptable in specified velocity that has been accomplished here. Also, RMS's of objective function before and after optimization are represented in Table 5 for better numerical comparison. The RMS of vertical acceleration and pitch angle of common and optimized suspension settings at different speeds is shown in Fig. 9. As it was expected the RMS of acceleration after optimization is less than common case within the range of 60 to 90 Km/hr, in which the problem was designed for. Beyond this point the changes in suspension system are not performing well, since a_{RMS} that

can be defined as $\left(\frac{a_{common} - a_{optimized}}{a_{common}} \right) \times 100$ is higher than common as it shown in Fig. 9a. As an example, at a velocity of 110 Km/hr optimum is 11.44% more than that of common system that means the optimization has not good performance if the designer does not consider the whole

range in the time of optimality, so one must precisely choose the range of optimization. Meanwhile, the reduction percentage of due to suspension modification is plotted vs. velocity in Fig. 10a which shows a declining trend. It means that the optimal design works best in the 60 km/hr and has the worst case in 110 Km/hr. Pitch angle has decreased substantially in and out of velocity range of 60 to 90 Km/hr. on contrary to , meaning pitch angle of optimized system is still 24.96% less than normal one, in $v=110$ Km/hr.

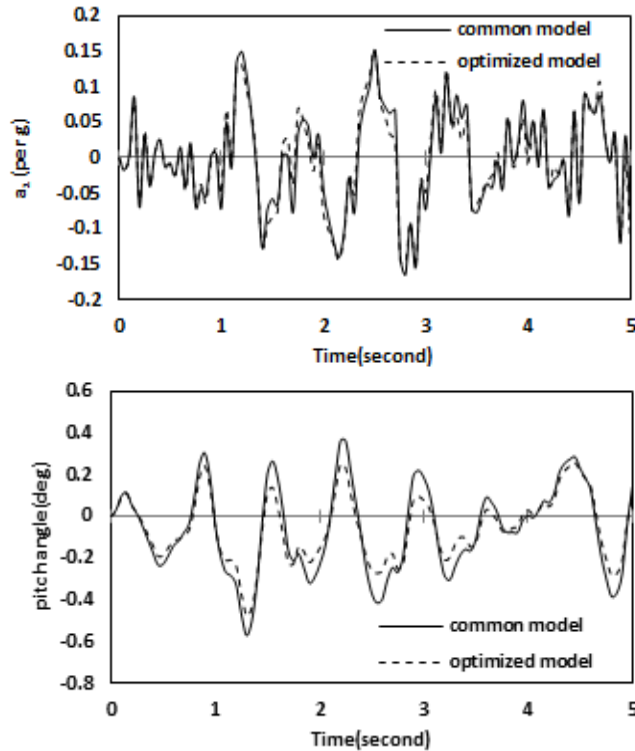


Fig. 8. Effect of spring and damper characteristics on ride quality

TABLE V
VERTICAL ACCELERATION AND PITCH ANGLE AFTER OPTIMIZATION

	Velocity (Km/hr)					
RMS	60	70	80	90	100	110
$a_{z, com.} (0.01g)$	4.4	4.69	6.11	6.97	8.30	7.78
$a_{z, opt.} (0.01g)$	4.13	4.54	5.90	6.85	8.25	8.67
Common θ (deg)	0.180	0.178	0.189	0.218	0.238	0.244
Optimized θ (deg)	0.172	0.164	0.162	0.168	0.177	0.183

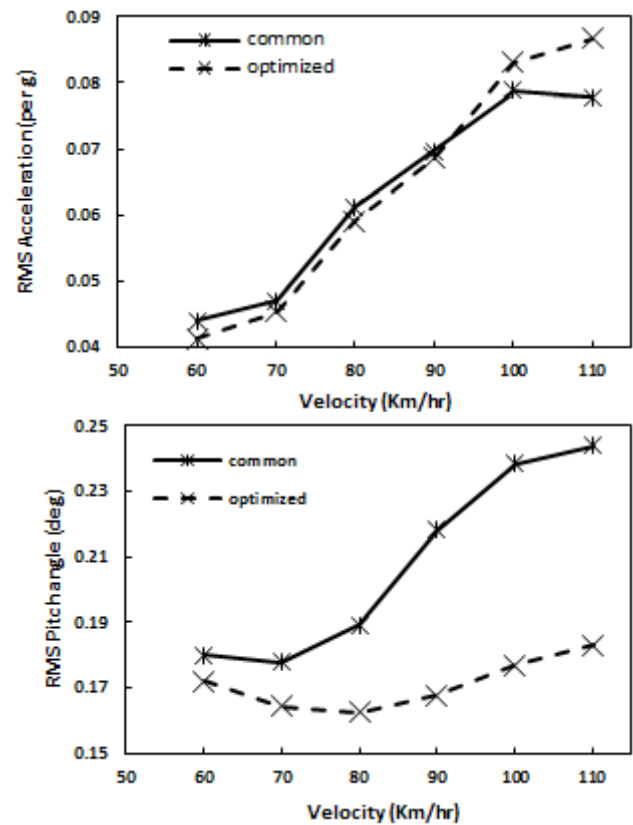


Fig. 9. RMS vertical acceleration and pitch angle for optimized and common suspension settings at different speeds

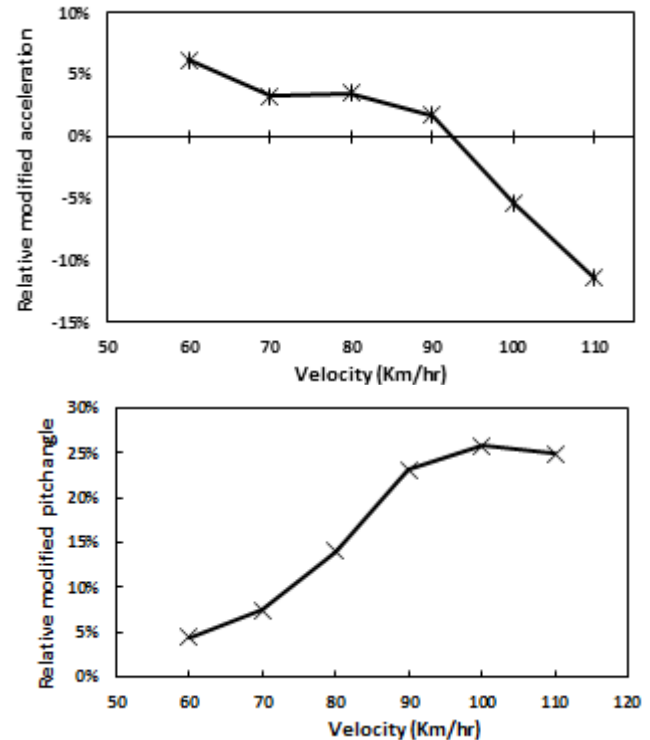


Fig. 10. Relative RMS vertical acceleration and pitch angle at different speeds

CONCLUSIONS

Conventional analytical studies on vehicle ride comfort in the low-frequency range are focused on the bounce and pitch behavior of vehicles. Due to high cost of active and semi-active suspension systems, passive ones can be used to determine proper stiffness coefficient of spring and damper in order to achieve the best ride comfort along with lowest expenses. DOE, as an efficient way of modeled parameter optimization using statistical method with saving cost and time, is a good approach for improving ride comfort. In this study, DOE has been applied on the results of a 7DOF model excited with a standard road to select F/R spring and damper coefficient resulting in minimum RMS of vertical acceleration and pitch angle for speed between 60 to 90 Km/hr. The optimal designed method shows that the vertical acceleration works best in the 60 Km/hr and will decrease the performance with increasing the automotive velocity, although the performance of pitch angle will improve from 5% to 25% with increasing the automotive velocity from 60 to 110 Km/hr.

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